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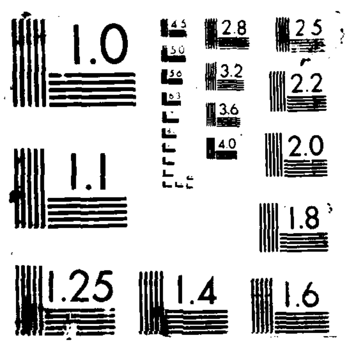
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RESEARCH ON A MINI-DISC MACHINE FOR GEAR TESTING

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THE COMMISSIONING AND USE OF A MINI-DISC SCUFFING RIG EMPLOYING RELATIVELY INEXPENSIVE TEST SPECIMENS IS DESCRIBED TOGETHER WITH PRELIMINARY SCUFFING TEST RESULTS ON THREE CANDIDATE OILS. RESULTS ARE COMPARED WITH AVAILABLE RYDER RESULTS.

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1.0 INTRODUCTION

The need for an alternative to the Ryder test has long been recognised. Repeatability and reproducibility leave a lot to be desired. However a considerable database of experience has been built up using the Ryder test and any replacement proposed must prove itself superior in its approach and suitability for routine evaluation work and, where possible, draw from the previous experience gained from Ryder tests. The USAF, aware of the potential of an alternative test, initiated various studies from an extension of the existing Ryder technique to the development of a disc machine test aimed specifically at oil evaluation.

As outlined in the proposal for this work (1) the advantages of disc testing over gear testing lies in the ability to vary the parameters considered to most affect the onset of scuffing, namely; contact stress, surface speed and slide/roll ratio. In addition close control is attainable over the material metallurgy and surface finish. Work at Imperial College by Macpherson (2) indicated the potential of the mini-disc particularly in terms of a reduced specimen cost. More recently however, its suitability to new material-developments, notably ceramics, has become apparent.

The objective of this study's first year's work was to commission and prove the mini-disc test machine, show it to be a practical tool, inexpensive to operate for general oil evaluation, and to demonstrate its potential to simulate Ryder machine conditions so that the existing data base of Ryder results are, in principle, compatible with future test results.

2.0 RESULTS AND DISCUSSION

2.1 TEST RIG

2.1.1 Operating Parameters

Merritt (3) first recorded that a pair of discs could reproduce contact conditions at any particular position of a gear's meshing cycle if the discs are rotated at the same angular velocities as the pinion and gear respectively, and the disc radii are chosen to equal those of the gear tooth profiles at the point of contact. Thus the relative radius of curvature of the two discs is the same as that of the pinion and gear. This simulation of the gear kinematics is incomplete insofar as contact conditions vary during the meshing cycle, whilst the relative surface motion of the discs remains constant. Nonetheless a very much closer control over operating conditions is possible with rotating discs.

Numerous empirical criteria have been proposed for scuffing of which the Blok postulate is the most well known; namely that the total contact temperature at which scuffing occurs is constant for a given combination of (non-additive) lubricant and rubbing surfaces. This has subsequently been extended to cover additive oils where the critical temperature is held from disc machine experience to correlate usefully with other parameters (4). However the calculation of the total contact temperature requires knowledge of the heat transfer coefficient, and of the instantaneous coefficient of friction, both of which are extremely hard to quantify accurately. Using the computer simulation "TELSGE" (5) total contact temperatures for the Ryder gears have been calculated for the three test lubricants. Only the results for Mil-L-23699 are quoted as they are similar bearing in mind the estimates made for the pressure viscosity coefficient (alpha), the temperature viscosity coefficient and the thermal conductivity. It should be stressed that these are simulated, for example it can be shown that a 50% increase in the alpha value doubles the minimum film thickness and raises the bulk temperature by 25%. With reference to Table I, Mil-l-23699, the minimum film thickness is calculated as 0.69 µm (2.8E-06"), and the flash temperature to be 271°C (520°F) for an ambient temperature of 175°F, similar to those calculated for the mini-disc. A number of other temperature criteria have been proposed confirming the conflict of evidence over the applicability of Blok's postulate, however in the absence of something better the Blok criterion is widely used. Pre-dating the criterion of total contact temperature is that of frictional power intensity; *the product of sliding speed and mean* Hertzian stress. This is often multiplied by the coefficient of friction to give a criterion of greater physical meaning; mean shear stress multiplied by sliding speed. By a similar argument to that used for the total contact temperature, the friction power intensity increases with increasing speed at a constant slide/roll ratio. However as with Blok's criterion an assumption is made of the coefficient of friction. A comparison of frictional power intensity between Ryder and mini-disc is as follows.

Power intensity criterion ($P_{mean} * V_{slid}$)

The mean Hertzian pressure (P_{mean}), the Hertzian half width (b), and the sliding velocity (V_{Slid}) are defined below. The numerical values of the relevant variables applicable to the Ryder and mini-disc tests are shown opposite.

E^1 =228.0 E+9 N/m^2 (33.1 E+6lbf/in^2)

$\omega = 2*\pi*RPM / 60$

W=Load (N)

Contact length

$V_{Slid} = \omega r_1 - \omega r_2$

L =4.76 E-3 m (mini-disc) (0.188 in)

$b = 8WR1/(\pi*E1*L)^{1/2}$

L =6.35 E-3 m (Ryder) (0.250 in)

$P_{mean} = W / (2*b*L)$

Equivalent radius

R^1 =8.26 E-3 m (Ryder 2nd ctct) (0.325 in)

R^1 =7.32 E-3 m (Ryder tip) (0.288 in)

R^1 =4.76 E-3 m (mini-disc) (0.188 in)

Using this data the following comparison can be made,

Ryder

2nd change point; (S/R =0.27) Power intensity criterion =124E+06 $*(W)^{1/2}$

tip; (S/R =0.37) Power intensity criterion =290E+06 $*(W)^{1/2}$

Mini-Disc

(S/R =0.27) Power intensity criterion =119E+06 $*(W)^{1/2}$

(S/R =0.37) Power intensity criterion =151E+06 $*(W)^{1/2}$

Although ultimate failure results when boundary protection is lost, any gear scuffing failure initially requires that there is a failure in the EHD film. From experience a λ value greater than 3 precludes any likelihood of scuffing, but a λ value of less than 3 does not provide any indication of the likely onset of scuffing. Whilst this is more likely due to an inadequate measure of surface roughness rather than any intrinsic failure in the criterion, it limits the applicability of λ values as a reliable design criterion. Table I shows that both the Ryder and mini-disc machines operate at λ levels well below 1 for much of the test period.

Experience shows that scuffing most frequently starts at the second change point. This was therefore the part of the meshing cycle chosen for simulation; i.e. where the reduced radius (R^1) = 8.26 mm and the slide roll ratio (S/R) = 0.27. The present design of the mini-disc rig has a fixed specimen centre distance which at 19.0 mm, gives a conforming contact of smaller reduced radius 4.76 mm.[1]

[1] The effect of this radius difference is to reduce the mini-disc theoretical smooth surface contact width by about 24%. This is not thought to be significant, and particularly when variations in contact width of this order and more are seen with rough surfaces in Hertzian contacts (Johnson 1985). For example a 1 μm rms roughness at typical scuffing loads on the mini-disc machine would be expected to result in a contact width about 25% bigger than the Hertzian value.

Table 1 OPERATING PARAMETERS

1			Mil-L-23699	
2				
3		Run-in load	Fail-load	Fail-load
4		(min alpha)	(min alpha)	(max alpha)
5	u1, in/s	175.6	175.6	175.6
6	u2, in/s	80.7	80.7	80.7
7	r1, in	0.375	0.375	0.375
8	r2, in	0.375	0.375	0.375
9	Viscosity, lb sec/in^2	1.13E-06	1.13E-06	1.13E-06
10	E1, psi	3.00E+07	3.00E+07	3.00E+07
11	E2, psi	3.00E+07	3.00E+07	3.00E+07
12	Poisson ratio 1	0.3	0.3	0.3
13	Poisson ratio 2	0.3	0.3	0.3
14	Alpha (estimates...), in^2/lb	7.50E-05	7.50E-05	1.50E-04
15	Facewidth, in	0.1875	0.1875	0.1875
16	Total load, lbf	40	2000	2000
17	Mean surface speed, in/sec	128.15	128.15	128.15
18	Surface rms, micro in	2.00E-05	2.00E-05	2.00E-05
19	Equivalent radius	0.1875	0.1875	0.1875
20	Equivalent modulus	32967033	32967032.97	32967032.97
21	Thermal conduct., lbf/°F/sec	6.49	6.49	6.49
22	Spec. heat, lbf.in/°F/sec	1026	1026	1026
23	Density, lbm/in^3	0.283	0.283	0.283
24	Inst. coeff. friction	0.01	0.05	0.05
25				
26	Speed parameter	2.3427E-11	2.3427E-11	2.3427E-11
27	Lubricant parameter	2472.52747	2472.527473	4945.054945
28	Load parameter	3.4513E-05	0.00172563	0.00172563
29	Film parameter	2.48E-05	1.49E-05	2.17E-05
30				
31	Min film thickness	4.65E-06	2.80E-06	4.07E-06
32				
33	Lambda	0.23	0.14	0.20
34				
35				
36	Hertz ctct width, in	0.00175779	0.012429439	0.012429439
37	Max ctct stress, psi	7.73E+04	5.46E+05	5.46E+05
38				
39	Flash temp. °F	5.55	522.08	522.08

Note:"SPEED", "LOAD", & "LUBRICANT" parameters as per Dowson and Higginsons' equation for line contacts.

For a pair of gears R^1 varies along the contact path between a maximum at the pitch point and a minimum at the tip and root. The slide/roll ratio varies in the opposite sense. Thus in the mini-disc machine where the specimens are of equal size, the discs are run at different speeds to achieve the required slide/roll ratio.

The mini-disc rig was modified to permit a maximum speed of 5000 rpm producing a maximum sliding-speed of 2 m/s (6.6 ft/s) at a slide/roll ratio of 0.27 ($^S/_R$ at 2nd change point) as compared with a sliding speed of 6m/s (19.7 ft/s) for the Ryder, or a sliding-speed of 2.7 m/s (8.9 ft/s) at a slide roll ratio of 0.37 ($^S/_R$ at gear tip) compared with the speed of 13.3 m/s (43.7 ft/s) for the Ryder.

It is unlikely that sliding speed alone is important if a thermal failure criterion, such as Blok's, is responsible for scuffing onset. Instead scuffing is more likely to depend upon the product of sliding speed and load (ie. a power intensity argument relating to frictional heating). Thus although the present design[2] of a mini-disc will not simulate the Ryder sliding speed conditions, the equivalent energy balance can be satisfied.

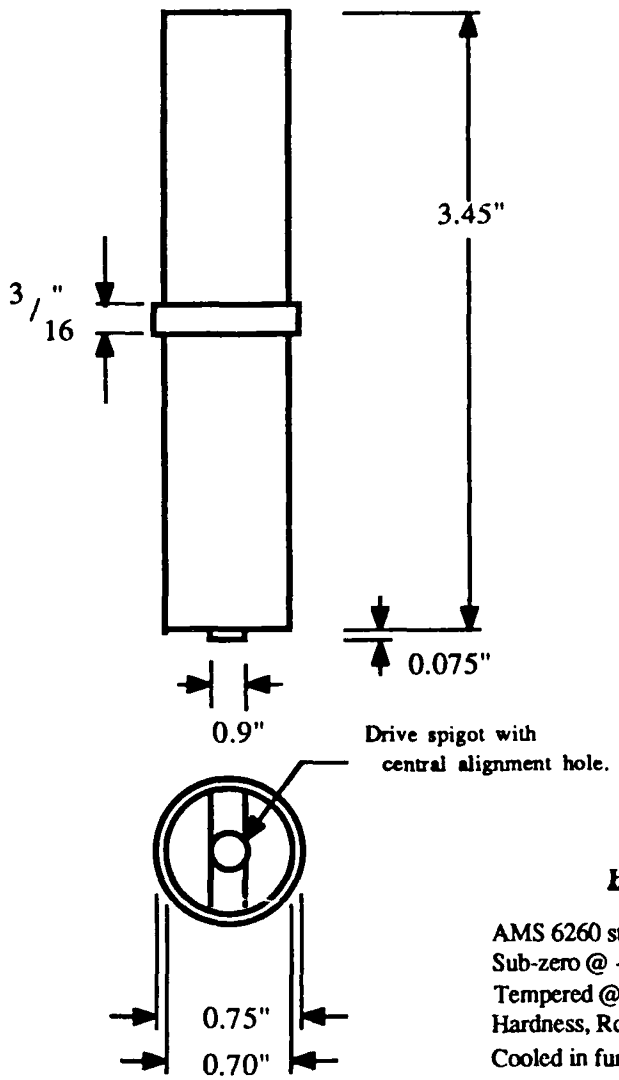
The surface finish of the gears is specified as 18-25 μins rms. The grinding direction is perpendicular to the direction of rolling whilst that of the discs is parallel. The importance of the direction of grinding has already been demonstrated by De Silva et al (6) and Johnson et al (7). Work by Staph, Ku and Carper (8) shows the beneficial influence of axial grinding in delaying the onset of scuffing however results for both axial and circumferentially ground discs lie on the same operating line of the m/c.

2.1.2 Specimen design

Mini-disc specimens are rod-like, the thicker central section comprises the test "disc". A spigot on the bottom face mates with a "universal" drive coupling, (Fig. 1). The sole difference between the two test specimens is that one has a narrower track of 4.76 mm ($^3/_{16}$") as against 12.7 mm ($^1/_2$") for the other. This reduces end effects and, for the purpose of calculating the contact stress, one considers the narrow track. The narrow tracked specimen is that driven at the higher rotational speed (V_1).

The simplicity of specimen design ensures close control over manufacture can be maintained particularly with regard to the surface finish. Although disc testing has in the past been carried out with "axially" ground specimens, the true direction of grinding is partly helical due to the feed. Axially ground specimens have not been attempted in this series of tests, however recent developments in grinding by RAE Farnborough encourage further work with axially ground specimens.

[2] It would not be difficult in a future version of mini-disc to design to achieve Ryder sliding speed similarity by increasing disc size or rotational speed. However it would be sensible to consider the results of these and future tests carefully before making such a change.



SURFACE FINISH

Circumferentially ground,
18-25 micro-inches

HEAT TREATMENT.

AMS 6260 steel gas carburised @ 620 degrees C
Sub-zero @ -70 degrees C
Tempered @140 degrees C for 4 hrs.
Hardness, Rc 59/62, case depth 0.032 in.
Cooled in furnace to 850 degrees C & oil quenched.

Mini-disc specimen, narrow track.

Figure 1.

The aim in the present tests has been to produce a surface finish with similar height and spatial characteristics to the Ryder gears. Carper & Ku (9) used a measure of composite surface roughness; $\delta' = (\delta_t + \delta_c)/2$ where δ_t is the transverse surface roughness and δ_c is the circumferential. This approach attempts to match anisotropic topography to that of the gears. In practice this is difficult to achieve and the surface finish of the discs used is similar to that specified for the Ryder gears albeit the grinding direction is different.

The specimen material is of the same specification as the Ryder gears; AMS 6260, heat treated to give a case hardness of 81/84 Rockwell A and a case depth of 0.81 mm (0.032"). Hardness checks were made on 1 per 4 test pieces, and case depth on 1 per 40.

2.1.3 Rig details

The normal method used to achieve a series of slide/roll ratios to simulate different parts of the gear tooth is by the use of a single motor driving both discs through belts and pulleys. By changing the pulley ratios relative speeds are obtained. This limits the different slide/roll ratios that can be obtained. The mini-disc machine can achieve a continuous range of ratios via the separate motors driving each disc. These are controlled by very stable thyristor converters with the input to one being a specified percentage of the other. Speed is regulated to within ± 0.1% of the maximum speed for 100% variation in load and is continuously variable between 0 and 5280 rpm. The slide/roll ratio may be varied up to maximum $^S/_R$ of 0.53.[3]

The specimens were originally supported and loaded by hydrostatic pads but misalignment between the discs required a modification to the rig design. Surfaces running in nominal line contact are very sensitive to edge effects, Bell found that an imbalance of less than 2% was sufficient to cause the majority of scuffs to occur at the more heavily loaded edge. Alignment was achieved by using the hydrostatic pads solely to support the specimens whilst load is applied via a hydraulic load cell, (Figure 2). This arrangement proved successful and subsequent failures occurred across the entire facewidth.

The test lubricant is the same as that used to lubricate the rig and support the discs. A trailing thermocouple controls the supply temperature to ± 1 ½°C (±°F). Despite extensive lagging, the length of pipework involved means that the sump temperature is 10-15°C (18-27°F) above that of the head and introduces a considerable thermal lag when running up to operating temperature.

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[3]In attempts to generate scuffing with ETO25 at high slide/roll ratios and high loads, it was noticable that the high torques generated slowed the motors. This condition was not allowed to exist for long for fear of damage to the motors.

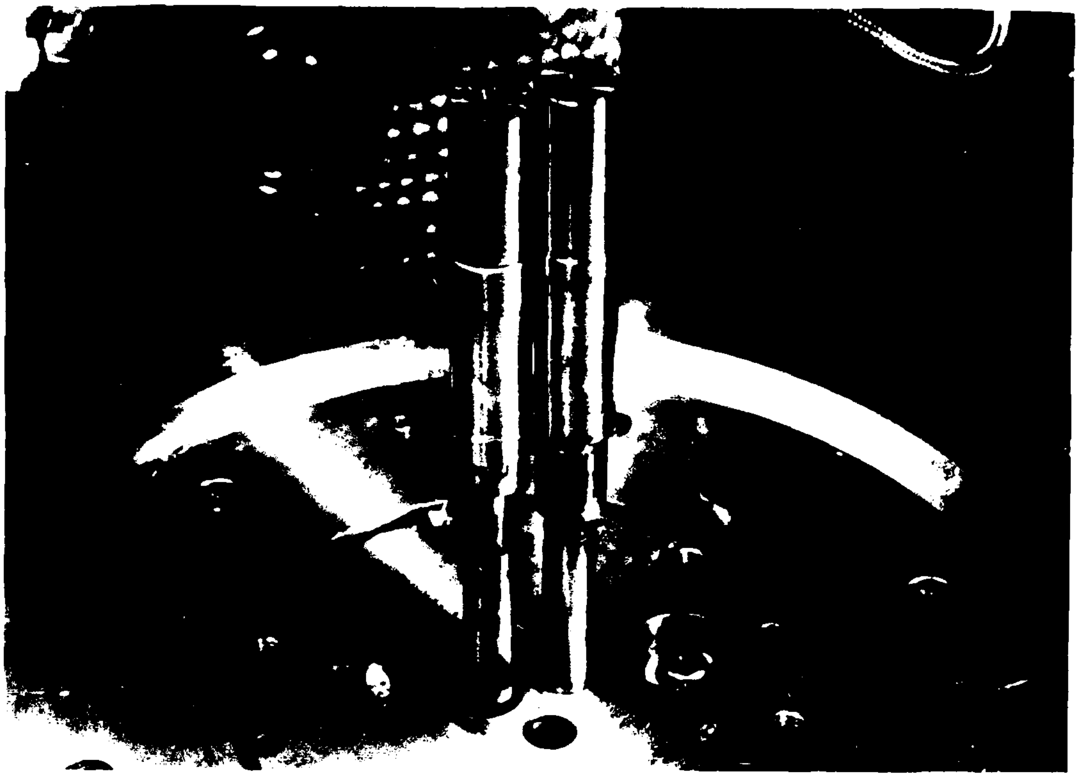
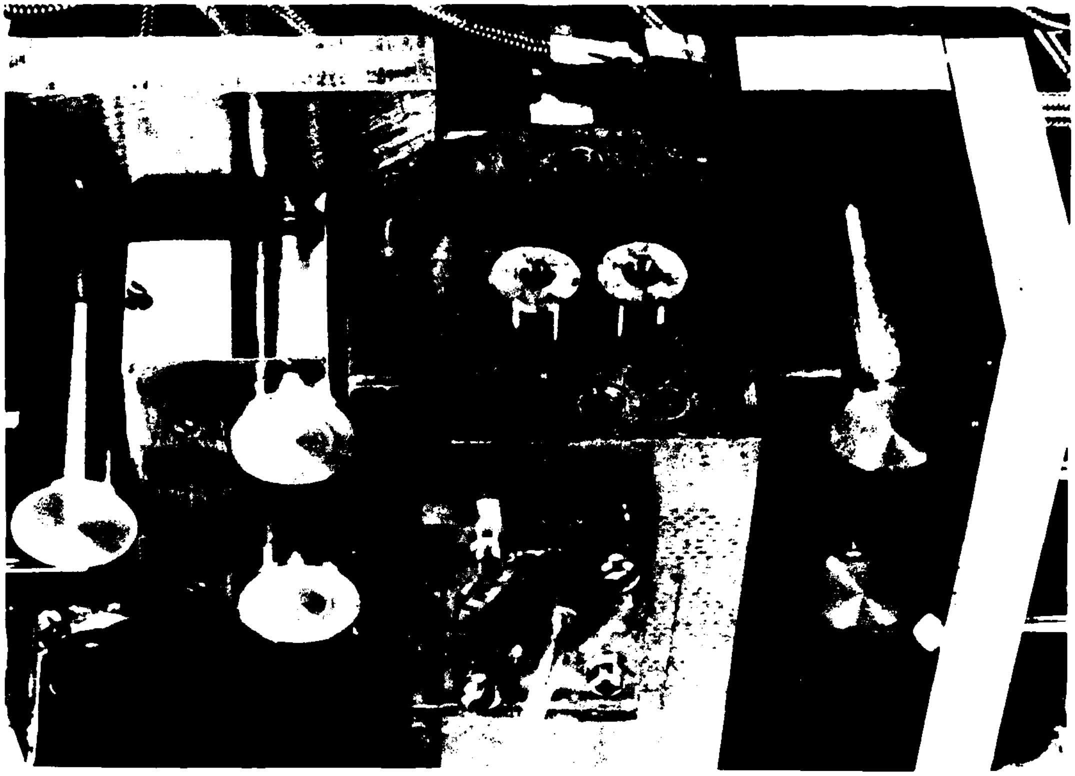


Figure 2 RIG LOADING ARRANGEMENT



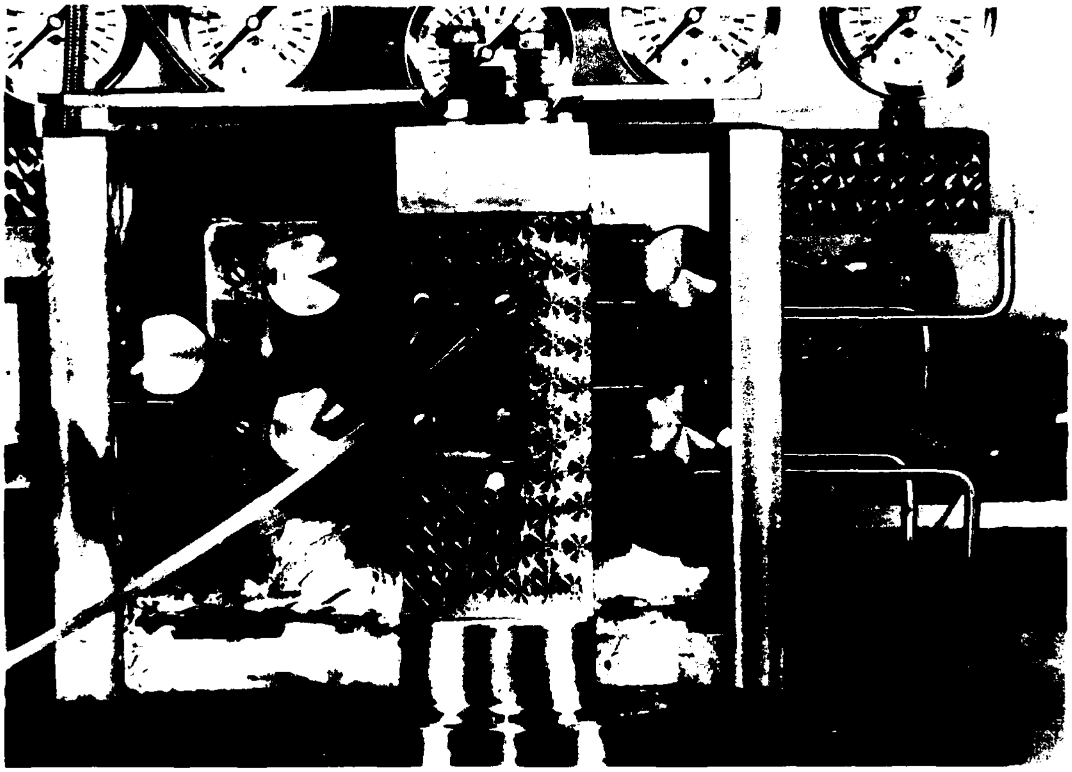


Figure 2 Continued

Marginal heating means that the cooling circuit is switched in and out as necessary whilst the heaters remain on constantly. Lubricant flowrate is is a minimum of 240 ml/min (0.06 US gal/min) compared with 270 ml/min for the Ryder.
An oil jet feeds the inlet and ensures that the contact remains fully flooded at all times. A statically calibrated torsion bar provides a measure of instantaneous torque which on occasion displays the classic scuff/recovery before final seizure.[4]

2.2 Test Method and Results

2.2.1 Instrumentation

The maximum disc speed is 5280 rpm, and the slide roll ratio infinitely variable between 0 and 0·53. The speed controller is calibrated to ± 2% of the indicated speed, and the speed variation is held to ± 0·1% by KTK 6P4Q thyristor control.

A strain gauged torsion bar forming a Wheatstone bridge provides a torque signal to both an LED indicator and a "Rikadenki" chart recorder. The latter is slightly damped to reduce excessive oscillations of the pen, thus reducing the sensitivity slightly.

An "Esher" thermostat controls the sump temperature according to that sensed by the inlet thermocouple. Inlet temperature is controlled within ±2°C (± 4°F) of that specified. The marginal heating means that operation above 80°C (183°F) requires running under load. The temperature is recorded synchronously with the torque on the chart recorder The recorder is run continuously, even during warm-up to provide evidence of exposure-time.

2.2.2 Rig operation and calibration

Before each test oil is run the rig is partially stripped and cleaned thoroughly in solvent, the pump heads being left intact. On reassembly and after flushing the first pass of the test oil is dumped.

A dummy set of specimens are mounted in the rig and run until the operating temperature is reached and has stablised. A load of about 3.5 bar (50 psi) is usually sufficient for this. An operating temperature of 84°C (183°F) is used for the bulk of the testing. At this stage the test specimens are inserted and the chart recorder started. Load is increased incrementally in 50 psi steps,each stage being run for a period of 3 minutes, until seizure. At seizure the drive motors invariably trip and require resetting before further running. Particular comments upon the run, such as any vibration are noted beside the torque trace and are reproduced in the list of results (Table II*).

[4]A considerable variation in both mean and instantaneous scuffing torque is evident from tests. It has not yet been established whether this is due to the drive arrangement or is a real effect.

Table 2 MINI-DISC RESULTS

Speci -men	Oil*	Temp. -erature	S/R	V_1 rpm	Scuff lb/in	Comments
1	ETO25	74°C	0.27	1100	No	Snatching, no fail at 450 psi when pads touched specimen.
2	ETO25	74°C	0.27	1100	1890	Inadequate temp.ctrl, >85 °C, m/c tripped twice @ 650 psi before scuffing @450 psi.
3	ETO25	74°C	0.27	1100	No	Test aborted after 5 min. running in for Talysurf measurement.
4	ETO25	74°C	.27-.37	1100	No	Rattling, no scuff on inspection @ 850 psi, even when S/R increased.
5	ETO25	74°C	0.37	1100	2310	Scuffed at end (misalignment?) failed @ 550 psi.
6	ETO25	74°C	0.27	1100	No	M/c tripped at 550 psi, reset & S/R 0.37, m/c tripped @ 450 psi. S/R 0.27, temp 84 °C no effect, drive failure, no scuff.
7	Drum A	84°C	0.37	2500	2940	No scuff at 850 psi, m/c trip. V_1 increased to 2500 rpm, temp to 84 °C scuff at 650 psi i.e. plenty of opportunity to run in.
8	Drum A	84°C	0.37	2500	1680	Uniform scuff at 450 psi, over entire area; little running in.
9	Drum A	84°C	0.37	2500	2310	Drive snatching. Stopped 3 times expecting scuff fromhigh torque indicated, finally scuffed @ 550 psi.
10	Drum A	84°C	0.37	2500	2520	Torque varied considerably, 25 psi load increments tried twice as long to scuff at 600 psi.
11	Drum A	84°C	0.37	2500	No	2 runs up to 850 psi @ S/R 0.27; no scuff.

*** Drum A :-Mil-L-7808, Drum C :-Mil-L-23699B**

Table 2 (Continued)

Speci -men	Oil*	Temp. -erature	S/R	V_1 rpm	Scuff lb/in	Comments
12	Drum A	84°C	0.37	2500	No	Several runs, including at 0.37 S/R but no scuff.
13	Drum A	84°C	0.37	2500	3360	Long warm up under no load.
14	Drum A	84°C	0.37	2500	1260	Massive wear. No shoulder left, torque low through-out test. Hardness to spec.
15	Drum A	84°C	0.37	2500	2310	Long warm-up with specimen stationary.
16	Drum C	84°C	0.37	2500	3360	Long (10 min) run in under low load (50 psi.)
17	Drum C	84°C	0.37	2500	3570	Long run in at 50 psi scuffed at 850 psi.
18	Drum C	84°C	0.37	2500	1680	Very little running in.
19	Drum C	84°C	0.37	2500	2940?	No visible scuff despite indications to the contrary.
20	Drum C	84°C	0.37	2500	3360	Recovered several times before tripping m/c, interesting wear scar with O.K. centre section.
21	Drum C	84°C	0.37	2500	2730	High torque recorded during running.
22	Drum C	84°C	0.37	2500	No	Run aborted due to faulty temp control.
23	Drum C	84°C	0.37	2500	3360	Long run-in, high torque throughout run.
24	Drum C	84°C	0.37	2500	1890	Recovery attempts.
25	Drum C	84°C	0.37	2500	1680	Classic torque trace as per cone / cylinder test. [5]

*** Drum A : Mil-L-7808, Drum C : Mil-L-23699B**

[5] A 'classic 'cone/cylinder test exhibits increased torque with each load step until the first scuff occurrs at which point there is a massive increase in torque, followed, if the oil is doped, by recovery and accompaning reduction in torque almost to that measured before the scuff. This scuff/recovery may occur several times before failure.

Table 2 (Continued)

Speci -men	Oil*	Temp. -erature	S/R	V_1 rpm	Scuff lb/in	Comments
26	Drum C	84°C	0.37	2500	420	Very little running in.
27	Drum C	84°C	0.37	2500	1260	High torque level during running in @ 50 psi.
28	ETO25	84°C	0.53	2950	No	No scuff despite severity of conditions, first temp. then speed increase but not both.
29	ETO25	84°C	0.37	2500	3150	1/2 facewidth, 10 mins run-in.
30	ETO25	84°C	0.37	2500	No	1/2 facewidth, Loaded up to 950 psi .
31	ETO25	84°C	0.37	2500	No	1/2 facewidth, Loaded up to 950 psi .
32	ETO25	84°C	0.37	2500	No	1/2 facewidth, Loaded up to 950 psi .
33	ETO25	84°C	0.37	2500	No	1/2 facewidth, Loaded up to 950 psi .

*** Drum A : Mil-L-7808, Drum C : Mil-L-23699B**

2.2.3 *Results*

The raw results for tests on the 2 oils supplied (Mil-L-7808 and Mil-L-23699) and on a 3rd oil ETO25 are shown in Table 1 together with rig operating conditions and comments noted at the time of testing.

The column marked "Fail" gives the hydraulic load cell pressure in lbf/in^2 at scuffing failure and can be converted into a line load of lbf/in by multiplying by 4.2 (Table 2 compares these results in terms of a line load in lbf/in with the Ryder results we have available.)

In order to match Ryder conditions as close as possible, tests were carried out at a temperature of 84°C (183°F) and a slide/roll ratio of 0.37. Some tests with ETO25 were carried out at other conditions in attempts to create scuffing, which in keeping with previous experience, proved difficult with this oil.

2.3 Discussion of Results

Although scuffing was readily achieved with the 2 main candidate oils (Mil-L-7808 and Mil-L-23699) it proved difficult with the 3rd oil chosen (ETO25). Also in all cases a significant variation in scuffing load was evident in the tests (i.e. for Mil-L-23699 the standard deviation is about 30% of the mean load and for Mil-L-7808 about 20%).

This variation is undoubtedly due to the period of running-in necessary to achieve temperature stability with the rig in its present form. The resulting modifications to surface topography (10) and the build-up of relatively thick chemical boundary films (11) during this running-in process are well documented as having a strong influence on subsequent scuffing performance. The latter seems of particular importance with these results as all 3 oils tested can be defined as particularly chemically active, and with the vast majority of the results an increased scuffing load is associated with an extended period of running-in.

In case of the Ryder test, ASTM 1947 lays down a specific sequence of test procedures to be followed, and with the Ryder's timer control, such a carefully controlled test would undoubtably reduce the spread of the results. This kind of formal test procedure could be defined for the mini-disc machine, but with the vast majority of lubricants of interest being of a chemically active type, an important element of their anti-scuffing potential in real practical situations would be ignored in any such formal test.[6]

[6] In recommending further work on the mini-disc machine it would be benificial to have the ability to change sliding speed rather than load. This would allow the chemical reactivity aspects of lubricants to be included in the map of an oils scuffing performance, and this could be achieved easily with the mini-disc machine, but not with the Ryder.

Table 2 shows a comparison of the present mini-disc result with the small number of appropriate results we have available from the Ryder with the same test oils. The results are shown in terms of the line-load at the onset of scuffing.

Although a considerable variation in scuffing load is evident in these results, the overall mean values are within about 15-20% of the few Ryder values we have available.

In both comparisons shown (Mil-L-7808 and Mil-L-23699) the mean mini-disc scuffing loads are higher than for the Ryder. There are good reasons to expect this form of result, as firstly, the mini-discs self-aligning capacity[7] will create a more uniform loading and hence allow higher mean loads to be applied before scuffing initiates, and secondly, the hydrostatic supports of the mini-disc machine allow a degree of flexibility which will reduce the dynamic vibrational loading almost certainly present in a more rigid system such as a Ryder machine.

2.4 FURTHER WORK

The high-frequency reciprocating (HFR) rig produces repeatable tests because close control of chemical and mechanical conditions is maintained; a new lubricant sample is used each time and the heating rate is controlled. With disc rig tests some running in procedure with each test is virtually inevitable. The approach of incremental loading is a function of the machine's ability to absorb increases in friction without a rise in the bulk temperature. Whether scuffing occurs depends upon a variety of factors but the loading sequence is not the simplest parameter to control as it varies several parameters simultaneously, as would increasing temperature. More sensible approach would be to increase the sliding speed progressively providing a similar, but alternative, to the cone-on-cylinder type test.

3.0 CONCLUSIONS

Several modifications to the Imperial College prototype mini-disc scuffing rig have been carried out prior to performing the scuffing tests described, in an attempt to approach Ryder conditions as closely as possible. The many improvements made, particularly in the case of loading and instrumentation, have established this form of rig design as a very versatile tool for performing rapid and inexpensive scuffing tests on candidate lubricants, and specimen materials and conditions.

[7]One disadvantage in a self aligned system which creates scuffing over the full face width is that roughness just prior to scuffing cannot generally be measured from the failed specimens.

The present specimens cost about £12/pair ($18)*, and neglecting any specialised running-in procedure or lubricant conditioning, specimen assembly, scuffing test and specimen removal can now be performed in a matter of minutes.

Scuffing test results are presented for the 2 candidate oils supplied by WPAFB (Mil-L-7808 and Mil-L-23699) and for a 3rd oil ETO25. Scuffing was successfully achieved for oils Mil-L-7808 and Mil-L-23699, and generally across the full face of the disc as a result of rig modifications to achieve self alignment. With ETO25 consistent and reliable scuffing results could not be achieved within the present operating range of the rig despite various attempts at increasing the severity of running conditions.

Although the results for Mil-L-7808 and Mil-L-23699 show considerable scatter in failure load, the mean values are in reasonable agreement with the appropriate Ryder results we have available. In the case of both oils the mini-disc mean results show slightly higher scuffing loads than would be expected from the Ryder results, but this difference is in keeping with the mini-disc's operating conditions, where the load is more evenly distributed across the specimen and some flexibility is afforded by the hydrostatic bearings. Both of these conditions would tend to increase the scuffing threshold load.

The results presented illustrate the importance, in terms of eventual scuffing performance, of chemically active lubricants and surface modification during a running-in phase. To reduce possible scatter in results, further tests could be carried out in a more controlled manner and in keeping with the stringent operating criteria of the Ryder or any other machine of interests.

Such an approach would be clearly acceptable for comparison of test purposes, but undoubtably questionable as a realistic means of lubricant assessment in relation to expected performance under the broad range of conditions met in practice. This is particularly important with chemically active lubricants of the type tested here, and which appear to offer the best anti-scuffing performance.

From these considerations it is clear that a much more superior test, which could easily be performed with the mini-disc machine, might be to maintain a constant load, which could be changed over a series of tests and sweep through a range of sliding speeds thus allowing the onset of failure to be mapped as a load-v-time phenomenon, which appears most relevant to modern lubricants. Previous Ryder test results would not need to be discarded by such a test, they would simply be valid across a single section through such a map.

Table 3

Comparison between Ryder and mini-disc results

Ryder (Naval Air Prop. Lab.)		mini-disc (Imperial)	
	(Mil-L-23699B)		
		3360 lb/in	
		3570 lb/in	
		1680 lb/in	
		2940 lb/in	
"A" side	2152 lb/in	3360 lb/in	
		2730 lb/in	
"B" side	2196 lb/in	3360 lb/in	
		1890 lb/in	
		1680 lb/in	
		1260 lb/in	
mean:-	2174 lb/in	2583 lb/in	
	(Mil-L-7808)		
set 1			
"A" side	2244 lb/in	2940 lb/in	
"B" side	2071 lb/in	1680 lb/in	
		2520 lb/in	
set 2		3360 lb/in	
"A" side	2040 lb/in	(1260 lb/in)	result ignored (No. 14)
"B" side	2126 lb/in	2310 lb/in	
mean:-	2120 lb/in	2562 lb/in	
	(ETO 25)		
	3877lb/in.	No reliable disc failures at the test conditions of 84 °C and $^S/_R$ of 0.37	

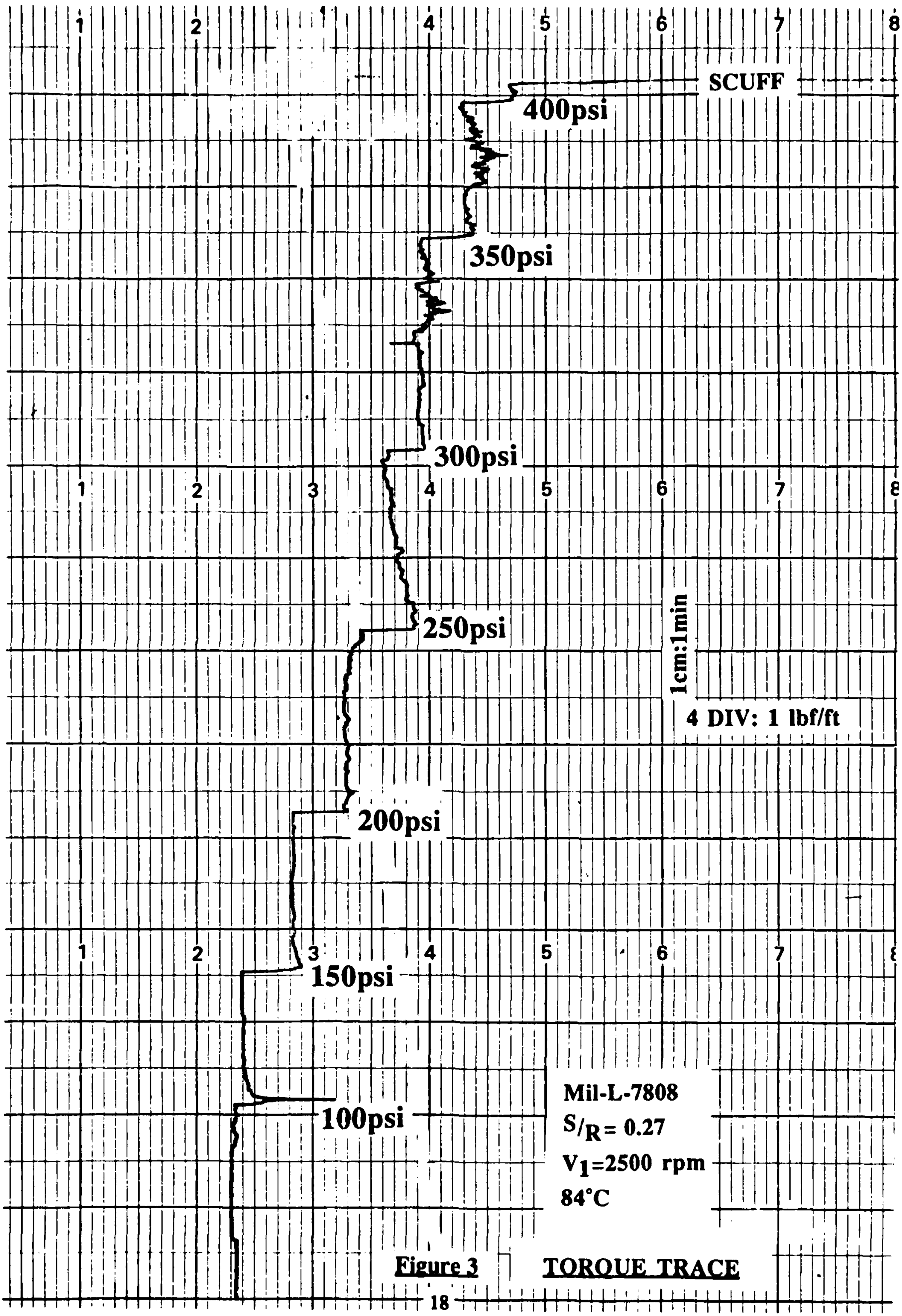


Figure 3 **TORQUE TRACE**

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